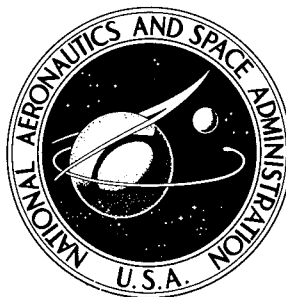


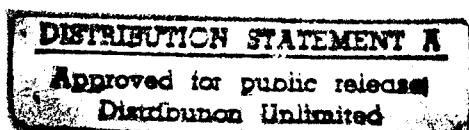
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FATIGUE LIFE OF
120-MM BORE BALL BEARINGS AT 600° F
WITH FLUOROCARBON, POLYPHENYL ETHER,
AND SYNTHETIC PARAFFINIC BASE LUBRICANTS

by

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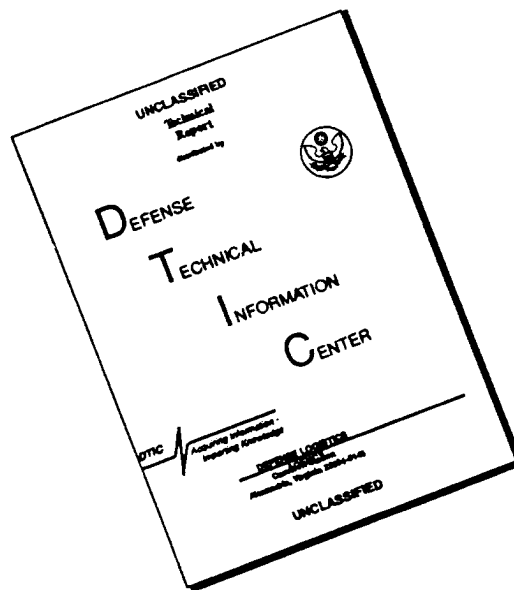
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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

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ABSTRACT

Groups of 120-mm bore angular-contact ball bearings made from AISI M-50 steel were fatigue tested at a temperature of 600° F (588 K) and a speed of 12 000 rpm with a synthetic paraffinic oil, a fluorocarbon, and a 5P4E polyphenyl ether. Under a low oxygen environment, the synthetic paraffinic oil and the fluorocarbon gave bearing lives approximately 14 and 3 times AFBMA-predicted (catalog) life, respectively. With the polyphenyl ether, bearing life was less than AFBMA-predicted (catalog) life in an air environment. For the synthetic paraffinic oil and the fluorocarbon, bearing fatigue was subsurface initiated. For the polyphenyl ether, bearing failure was mainly from surface distress, wear, and superficial pitting. (Author)

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SUMMARY

Groups of 120-mm bore angular-contact ball bearings made from AISI M-50 steel were fatigue tested with a synthetic paraffinic oil, a fluorocarbon, and a 5P4E polyphenyl ether. Test conditions included an outer-race temperature of 600° F (588 K), a speed of 12 000 rpm, and a thrust load of 4365 or 5800 pounds (19 416 or 25 800 N) producing a maximum Hertz stress of 295 000 or 323 000 psi (203 000 or 223 000 N/cm²), respectively, on the bearing inner race.

Under a low oxygen environment (less than 0.1 percent by volume) and at a maximum Hertz stress of 323 000 psi (223 000 N/cm²), bearings tested with the synthetic paraffinic oil had a life of approximately 14 times AFBMA-predicted (catalog) life. This fluid provided adequate elastohydrodynamic lubrication whereby no measurable wear occurred. The fluorocarbon fluid under the low oxygen environment and at a maximum Hertz stress of 295 000 psi (203 000 N/cm²) provided bearing life approximately three times AFBMA-predicted (catalog) life. This fluid did not provide consistent lubrication. The predominant lubrication mode ranged from boundary to elastohydrodynamic. In an air atmosphere with the 5P4E polyphenyl ether, component wear and surface distress occurred at a maximum Hertz stress of 295 000 psi (203 000 N/cm²), indicating predominantly boundary lubrication. The resulting fatigue life was less than AFBMA-predicted (catalog) life.

For the synthetic paraffinic oil and the fluorocarbon at 600° F (588 K), bearing fatigue was subsurface initiated. For the polyphenyl ether, failure resulted mainly from surface distress, wear, and superficial pitting.

*General Electric Company, Cincinnati, Ohio.

INTRODUCTION

A critical problem in the operation of supersonic and hypersonic aircraft will be the high ambient temperature to which the engine and gear box bearings will be exposed. Preliminary research reported in references 1 and 2 indicated that a synthetic paraffinic oil with an antiwear additive can give bearing fatigue life equivalent to AFBMA-predicted (catalog) life at temperatures between 550^o and 600^o F (560 and 588 K) under a low oxygen environment. Polyphenyl ethers had poor life potential; however, these tests indicated that a 5P4E polyphenyl ether with an oxidation inhibitor has the capability of operating in an air environment at temperatures to 600^o F (588 K). Ester-base lubricants under a low oxygen environment indicated a temperature limitation of approximately 500^o F (533 K) because of thermal degradation of the fluid. The ester-base fluids had a tendency to form hard coke deposits on the bearings at 500^o F (533 K) under a low oxygen environment.

Further work reported in reference 2 using a rolling-element fatigue tester indicated that in fatigue tests at 600^o F (588 K) the synthetic paraffinic oil exhibited a 10-percent fatigue life two and three times greater than the lives obtained with a fluorocarbon and the polyphenyl ether lubricants, respectively. This research also reported (ref. 2) that bearing torque decreased with outer-race temperatures up to 700^o F (644 K) for several chemical-base stocks. Decreasing torque was observed with decreased lubricant viscosity. Based on running track appearance, apparent elastohydrodynamic (EHD) lubrication existed when sufficient lubricant flow was maintained at a maximum Hertz stress of 192 000 and 165 000 psi (132 000 and 114 000 N/cm²) at the inner and outer races, respectively.

The objective of the research reported herein was (1) to establish the feasibility of operating large-diameter rolling-element bearings at 600^o F (588 K) under conditions of speed and maximum load approximating those experienced by a mainshaft bearing in jet engines, and (2) to determine in large-diameter rolling-element bearings the life expectancy with three advanced high-temperature lubricating fluids.

Tests were conducted in a high-temperature bearing tester at temperatures to 600^o F (588 K) with 120-mm bore angular-contact ball bearings made of consumable-electrode vacuum-melted (CVM) AISI M-50 steel having a Rockwell-C hardness in excess of 58 at a temperature of 600^o F (588 K). Test conditions included a speed of 12 000 rpm and bearing thrust loads of 4365 and 5800 pounds (19 416 and 25 800 N) which produced maximum Hertz stresses of approximately 295 000 and 323 000 psi (203 000 and 223 000 N/cm²), respectively, on the bearing inner race. Fatigue-life results were evaluated with respect to failure appearance, lubricant degradation, and deposit formation. All experimental results for a given lubricant were obtained from the same lubricant batch. Each component of the test bearings was from the same heat of material. All tests were conducted

at the General Electric Company, Cincinnati, Ohio, under contract to NASA and were initially reported in reference 3.

APPARATUS, SPECIMENS, AND PROCEDURE

High-Temperature Fatigue Tester

A schematic diagram of the high-temperature fatigue tester is shown in figure 1. In essence, the tester comprises two stationary housings connected by a bellows to form the bearing housing assembly. The rear housing assembly is fixed and contains an auxiliary roller bearing (not shown) to support a test shaft at the drive end, which thus leaves the forward housing assembly free floating.

The two test bearings are mounted on the test shaft and are separated by a shaft sleeve. The test bearings are retained on the shaft by a bolted retaining plate. The shaft is thus located by the test bearings and has axial freedom through the auxiliary roller bearing.

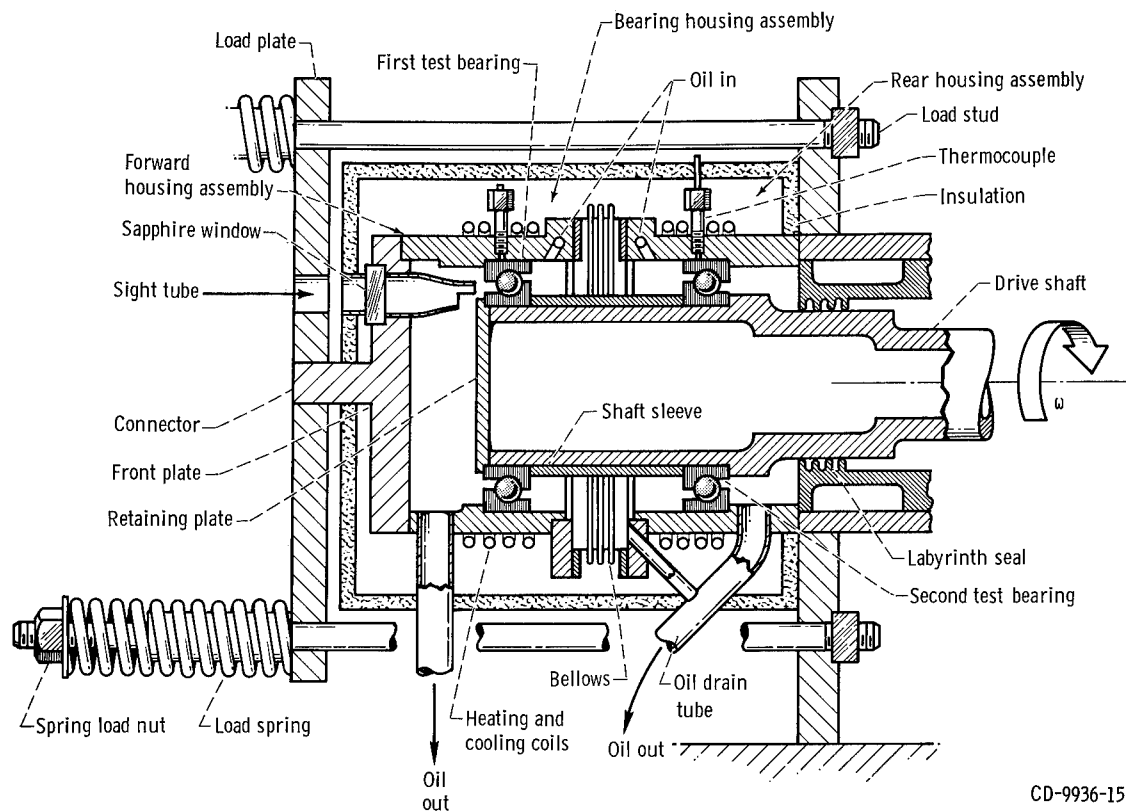


Figure 1. - High-temperature bearing fatigue test apparatus.

Loading is accomplished by a system containing 10 springs of 10 000 pounds (44 482 N) total working capacity. The two test bearings in the housing assembly are axially loaded against each other by the spring system. The load path is from the springs to a circular load plate through a connector to the front plate of the bearing housing assembly. The load is then transmitted to the outer race of the first test bearing and through the shaft sleeve to the inner race of the second test bearing.

Drive of the test rig is accomplished by a flat belt on a crowned spindle (not shown) at the auxiliary roller bearing end of the shaft.

Lubrication is provided to the test bearings through a jet-feed lubrication system by a pump immersed in a heated oil reservoir. The reservoir has approximately 3 gallons (11 400 cm³) capacity. The pump is capable of circulating the oil through the system at 3 gallons per minute (190 cm³/sec) at 600° F (588 K). Gravity drainage for the lubricant is provided by a single exit under each test bearing and also by the bellows in the bearing housing assembly.

Sealing of the oil in the bearing housing assembly from the drive system was provided by a labyrinth seal and a tandem carbon seal (not shown in the schematic diagram). The labyrinth seal was nitrogen pressurized to maintain a low oxygen atmosphere in the bearing housing assembly. The test shaft was driven at 12 000 rpm. Instrumentation provided for automatic shutoff by monitoring gas temperature, oil temperature, bearing vibration, and nitrogen dew point, flow rate, and pressure. If any of these parameters varied from those programmed for the test conditions, the test was shut down. Oxygen content within the bearing housing assembly was monitored during operation. An infrared pyrometer was used to measure inner-race temperatures through a sight tube aimed at the inner race of the first test bearing.

Test Procedure

The test bearings were installed in the test rig. The test load was applied and calibrated through a force gage attached to the load plate and the connector. (The gaging system is not shown in the schematic diagram.) After the load was applied, either nitrogen gas or air under pressure was supplied through a manifold system into the rig. For the nitrogen supply, the source was liquid nitrogen which was vaporized and heated through a heat exchanger. The lubricant in the oil reservoir was heated to 250° F (394 K) by use of a salt bath to achieve adequate pumpability. Heat generation from the test bearings was sufficient to bring the oil temperature up from the 250° F (394 K) starting level to 600° F (588 K). Control of the operating temperature was achieved by using an automatic water-oil heat exchanger. Test temperature stabilization was normally achieved within 1/2 hour after the test was started.

Test Bearings

The test bearings were ABEC-5 grade, split inner-race 120-mm bore angular-contact ball bearings having a nominal contact angle of 20° . The inner and outer races were manufactured from one heat of consumable-electrode vacuum-melted (CVM) AISI M-50 steel, and the balls were manufactured from a second heat. The nominal hardness of the balls and races was Rockwell C-63 at room temperature. Each bearing contained 15 balls 13/16 inch (2.0638 cm) in diameter. The cage was a one-piece outer-land riding type made of a nickel-base alloy (AMS 4892) having a nominal Rockwell-C hardness of 33. The retained austenite content of the ball and race material was less than 3 percent. The inner and outer race curvatures were 54 and 52 percent, respectively. All components with the exception of the cage were matched within ± 0.5 Rockwell-C point. This matching assured a nominal differential hardness in all bearings (i. e., the ball hardness minus the race hardness, commonly called ΔH) of zero.

A chemical analysis of the two heats of AISI M-50 material is given in table I. The M-50 has the capability of maintaining a Rockwell-C hardness of approximately 58 at a temperature of 600°F (588 K). This material is currently used by most jet engine manufacturers for high-temperature bearing applications.

TABLE I. - CHEMICAL ANALYSIS
OF CONSUMABLE-ELECTRODE
VACUUM-MELTED AISI M-50
BEARING STEEL

Element	Composition, wt. %
Carbon	0.802
Manganese	.24
Phosphorus	.008
Sulfur	.003
Silicon	.18
Chromium	3.95
Molybdenum	4.36
Vanadium	.93
Iron	Bal.

Test Lubricants

Three lubricants were evaluated with the 120-mm bore angular-contact ball bearings made of the AISI M-50 steel. These lubricants were a synthetic paraffinic oil with anti-wear and antifoam additives, a fluorocarbon-base fluid, and a 5P4E polyphenyl ether with an oxidation inhibitor and an antifoam additive. A standard ASTM chart showing the viscosity-temperature relation of these fluids is presented in figure 2.

Synthetic paraffinic oil. - Properties of the synthetic paraffinic oil are given in table II. This lubricant, a 100-percent paraffinic fluid with relatively high viscosity at high temperature (fig. 2), contained an antiwear additive and an antifoam agent. The addition of the antifoam agent was required when lubricant foaming occurred because of entrapped gases.

In order to prevent excessive oxidation of the fluid from occurring, it was necessary to maintain the fluid in a low oxygen environment (less than 0.1 percent oxygen by volume) at 600° F (588 K).

Fluorocarbon. - The properties of the fluorocarbon lubricant are also given in table II, and the temperature-viscosity characteristics are shown in figure 2. This lubricant is a polymeric perfluorinated high-temperature fluid having high oxidative and

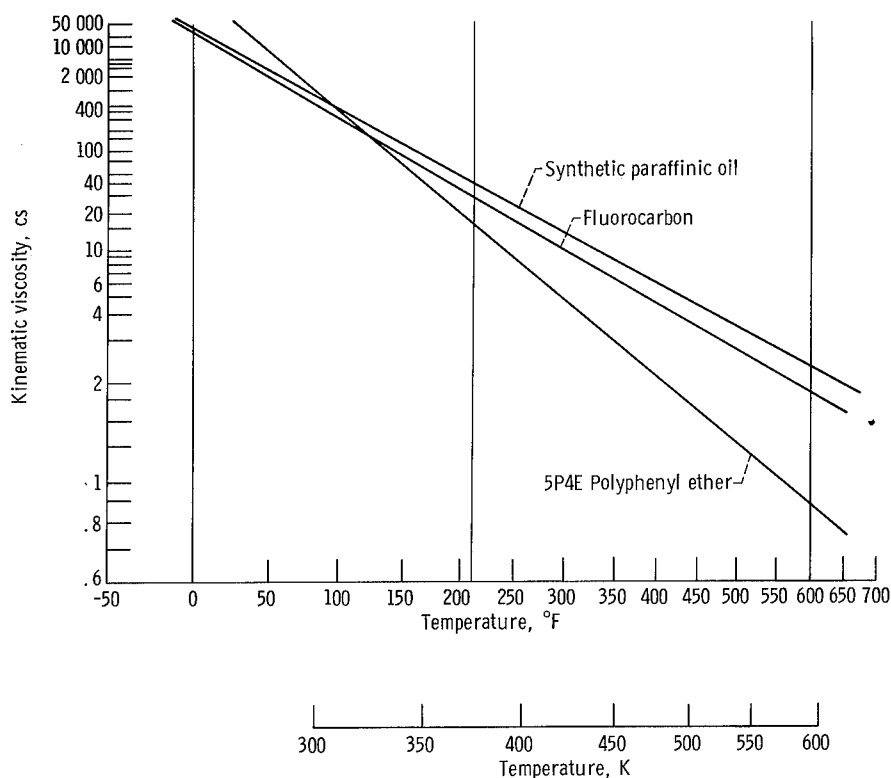


Figure 2. - Viscosity of experimental lubricants as function of temperature.

TABLE II. - TEST LUBRICANT PROPERTIES

Property	Lubricant description		
	Synthetic paraffinic oil	Polymeric perfluorinated fluid (fluorocarbon)	5P4E Polyphenyl ether
Additives	Antiwear agent Antifoam agent	None	Oxidation inhibitor Antifoam agent
Kinematic viscosity, cs, at - 100° F (311 K)	443.3	298.3	358.0
210° F (372 K)	39.7	29.8	13.0
400° F (478 K)	5.8	4.6	2.1
Flash point, °F (K)	515 (542)	None	5.40 (555)
Fire point, °F (K)	600 (588)	None	660 (622)
Autoignition temperature, °F (K)	805 (703)	None	1135 (886)
Pour point, °F (K)	-35 (236)	-30 (239)	40 (278)
Volatility (6.5 hr at 500° F (533 K)), wt. %	14.2	18.0	8.5
Specific heat at 500° F (533 K), Btu/(lb)(°F) (J/(kg)(K))	0.695 (2.91×10 ³)	0.316 (1.32×10 ³)	0.53 (2.22×10 ³)
Thermal conductivity at 500° F (533 K), Btu/(hr)(ft)(°F) (J/(m)(sec)(K))	70×10 ⁻³ (0.12)	51.5×10 ⁻³ (0.089)	78×10 ⁻³ (0.134)
Specific gravity at 500° F (533 K)	0.71	1.51	1.01

thermal stability (refs. 4 and 5). The fluid has a high density and poor heat-transfer characteristics (table II). These properties can result in higher bearing temperatures. At temperatures above 550° F (560 K), the fluid has a tendency to attack chemically some iron-base materials, as well as certain other alloy systems.

Tests were conducted with various types of test system materials at 600° F (588 K) in air to determine corrosion effects, if any. At 600° F (588 K) the major structural material, ASTM 4340, was badly corroded by the fluorocarbon lubricant. However, the same steel with a nominal 0.0005-inch (0.0013-cm) nickel plating showed no effects at all. AISI M-50 bearing steel and the nickel-base cage material showed good corrosion resistance with this fluid. From these tests, it was concluded that it is necessary to provide

nickel plating of exposed metal to inhibit any possible corrosion effects at 600° F (588 K).

Polyphenyl ether. - The properties of the 5P4E polyphenyl ether are given in table II and the temperature-viscosity characteristics are presented in figure 2.

Preliminary bearing tests with this fluid indicated that an antifoam agent was necessary for the same reasons discussed for the synthetic paraffinic oil.

RESULTS AND DISCUSSION

Groups of 120-mm bore angular-contact ball bearings made from consumable-electrode vacuum-melted (CVM) AISI M-50 steel were tested with three lubricants. These lubricants were a synthetic paraffinic oil having an antiwear additive and an anti-foam agent, a fluorocarbon having no additives, and a 5P4E polyphenyl ether having an oxidation inhibitor and an antifoam agent. Test conditions are given in table III.

TABLE III. - TEST CONDITIONS FOR 120-MM ANGULAR-CONTACT BALL BEARING
WITH THREE HIGH-TEMPERATURE LUBRICANTS

[Material, AISI M-50 steel; speed, 12 000 rpm.]

Lubricant	Bearing thrust load, lb (N)	Maximum Hertz stress, psi (N/cm ²)		Temperature, °F (K)			
		Inner race	Outer race	Inner race	Outer race	Oil in	Oil out
Synthetic paraffinic oil	5800 (25 800)	323 000 (223 000)	267 000 (184 000)	600 to 610 (588 to 594)	580 to 590 (577 to 583)	545 to 565 (558 to 564)	585 to 600 (580 to 588)
Polymeric perfluorinated fluid (fluorocarbon)	4365 (19 416)	295 000 (203 000)	248 000 (171 000)	600 to 610 (588 to 594)	580 to 590 (577 to 583)	525 to 560 (547 to 557)	590 to 615 (583 to 597)
5P4E Polyphenyl ether	4365 (19 416)	295 000 (203 000)	248 000 (171 000)	600 to 610 (588 to 594)	580 to 590 (577 to 583)	525 to 545 (546 to 558)	585 to 595 (580 to 586)

Synthetic Paraffinic Oil

Preliminary tests were run with the synthetic paraffinic oil with four bearings at a thrust load of 7000 pounds (31 138 N), which produced a maximum Hertz stress of 342 000 psi (236 000 N/cm²) on the inner race. Ball diametrical wear averaging 0.002 to 0.003 inch (0.0051 to 0.0076 cm) occurred in each bearing for operating times up to 165 hours (118×10⁶ inner-race revolutions). These results indicated that at this stress mixed lubrication occurs, that is, a combination of boundary and elastohydrodynamic (EHD) lubrication, with boundary lubrication being the predominant mode. As a result, the thrust load of 5800 pounds (25 800 N) was selected for fatigue tests to assure predominantly EHD lubrication. The fatigue-life results for the 26 bearings tested are shown in figure 3 and are summarized in table IV. The failure index (i. e., the number of fatigue failures out of the number tested) was 6 out of 26. The AFBMA-predicted (catalog) life at this load condition is also given for comparative purposes.

Typical fatigue spalls occurring on the balls of a bearing run with the synthetic paraffinic oil can be seen in figure 4. Metallurgical examination of the bearings indicated that they failed by classical rolling-element fatigue. The fatigue spalls were of subsurface origin, initiating in the zone of resolved maximum shearing stresses. An inner-race failure is shown in figure 5. A bearing run to suspension (500 hr of operation) is shown in figure 6.

Surface profile measurements of the bearing components made after testing, together

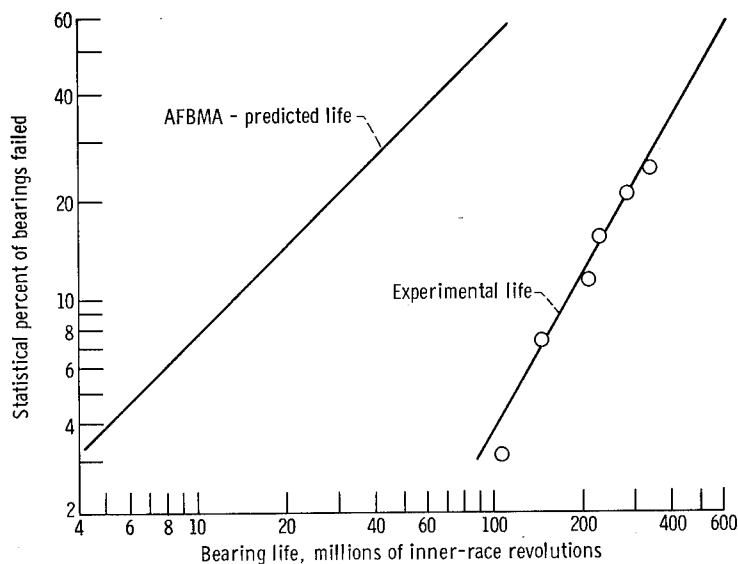


Figure 3. - Rolling-element fatigue life of 120-mm bore angular-contact ball bearings with synthetic paraffinic oil. Material, AISI M-50 steel; thrust load, 5800 pounds (25 800 N); speed, 12 000 rpm; temperature, 600° F (588 K); low oxygen environment; failure index, 6 out of 26.

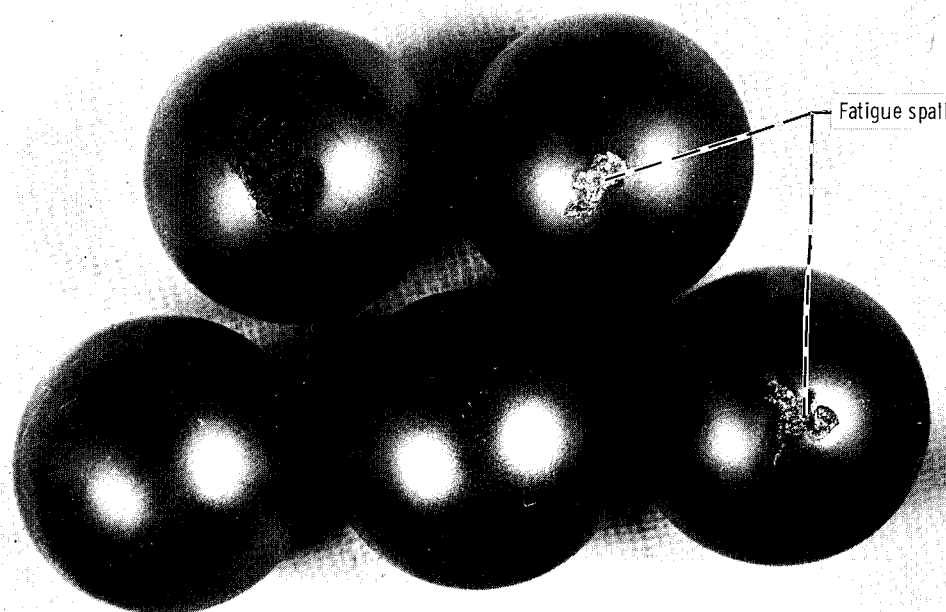
TABLE IV. - FATIGUE-LIFE RESULTS FOR 120-MM ANGULAR-CONTACT BALL

BEARINGS WITH THREE HIGH-TEMPERATURE LUBRICANTS

[Material, AISI M-50 steel; speed, 12 000 rpm; temperature, 600° F (588 K).]

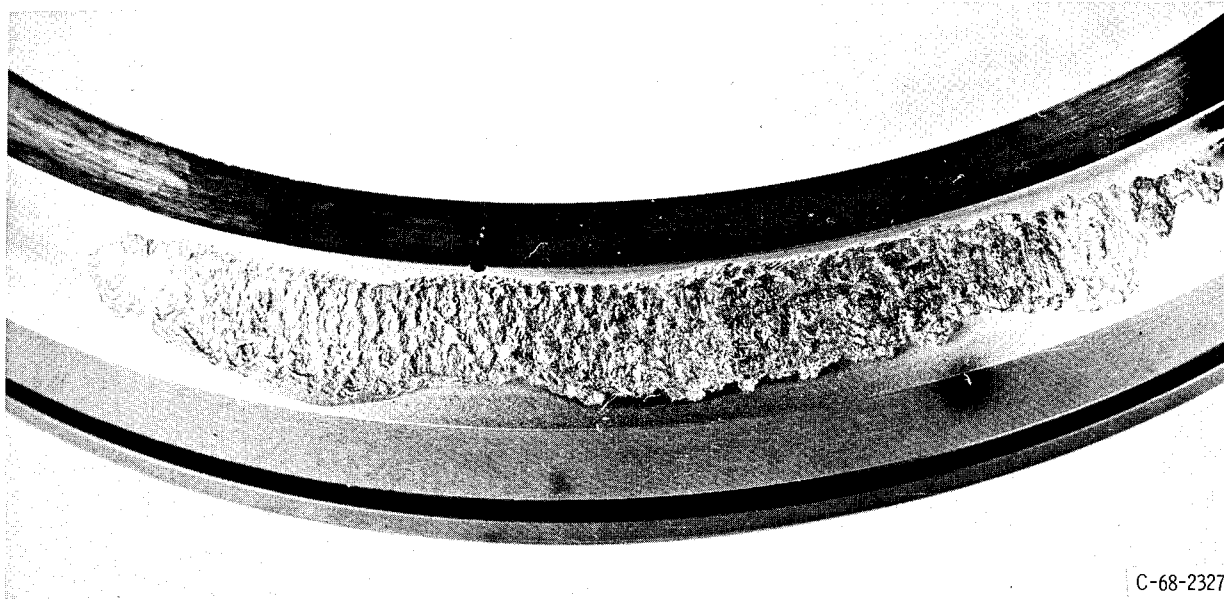
Lubricant	Additives	Bearing thrust load, lb (N)	Experimental life, millions of inner-race revolutions		Weibull slope	Failure index ^b	AFBMA-predicted 10-percent life, millions of inner-race revolutions	Ratio of experimental 10-percent life to AFBMA-predicted life
			10-Percent life	50-Percent life				
Synthetic paraffinic oil	Antiwear agent Antifoam agent	5800 (25 800)	182	512	1.8	6 out of 26	13.4	~13.6
Polymeric perfluorinated fluid (fluoro-carbon)	None	4365 (19 416)	92	357	1.4	4 out of 17	31.7	~3.0
5P4E Poly-phenyl ether	Oxidation inhibitor Antifoam agent	4365 (19 416)	^a 22	---	---	2 out of 26	31.7	~0.7

^aEstimated.^bNumber of fatigue failures out of number of bearings tested.



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Figure 4. - Typical fatigue spalls on bearing balls run with synthetic paraffinic oil. Material, AISI M-50 steel; thrust load, 5800 pounds (25 800 N); speed, 12 000 rpm; temperature, 600° F (588 K); low-oxygen environment; running time, 375 hours (270×10^6 inner-race revolutions).



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Figure 5. - Fatigue failure on bearing inner race run with synthetic paraffinic oil. Material, AISI M-50 steel; thrust load, 5800 pounds (25 800 N); speed, 12 000 rpm; temperature, 600° F (588 K); low-oxygen environment; running time, 189 hours (136×10^6 inner-race revolutions).



C-68-2328

Figure 6. - Unfailed 120-mm angular-contact ball bearing run with synthetic paraffinic oil. Material, AISI M-50 steel; thrust load, 5800 pounds (25 800 N); speed, 12 000 rpm; temperature, 600° F (588 K); low-oxygen environment; running time, 500 hours (360x10⁶ inner-race revolutions).

with pretest and post-test weight measurements, indicated that there was essentially no measurable wear or weight change in the bearing components with the synthetic paraffinic oil. On the basis of these surface trace measurements together with the appearance of the rolling-element surfaces, it can be concluded that almost complete EHD lubrication was obtained throughout the course of testing these bearings. Some of the bearings tested, however, had a slight glazed appearance, which indicated some asperity contact.

Viscosity measurements were made with the synthetic paraffinic lubricant at 100° F (311 K) as a function of bearing operating time. The fluid was circulated through each bearing at the rate of 1½ gallons per minute (95 cm³/sec). A representative plot of viscosity at 100° F (311 K) as a function of time is shown in figure 7. Initially, the viscosity decreased with time. The viscosity of the fluid then increased with time to slightly above its initial value. The neutralization number was essentially unchanged during the course of a test.

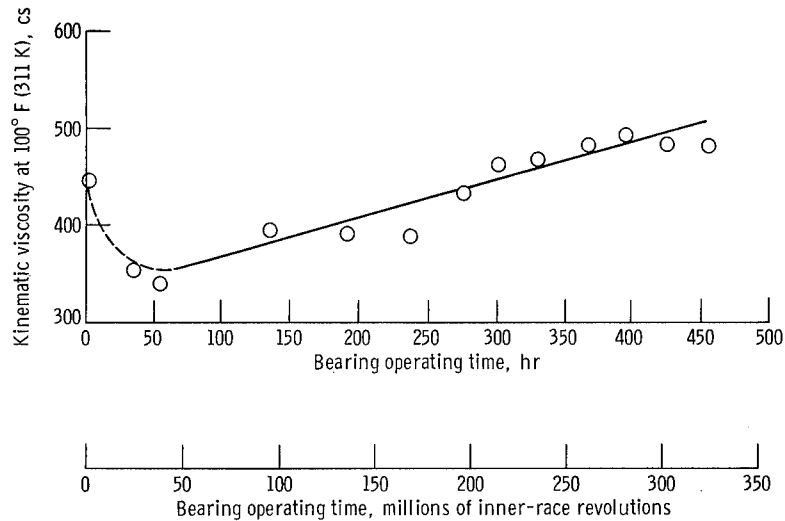


Figure 7. - Effect of thermal exposure and mechanical stressing on viscosity of synthetic paraffinic oil. Bearing operating temperature, 600° F (588 K); low oxygen environment; lubricant flow rate through bearing, $1\frac{1}{2}$ gallons per minute (95 cm³/sec); no makeup fluid added.

Although some discoloration was observed in the oil after test, these visual observations tend to indicate that no significant oxidation or thermal breakdown of the oil occurred at the test temperature under the low oxygen environment (less than 0.1 percent by volume).

Fluorocarbon Oil

Preliminary tests were conducted using eight bearings with the fluorocarbon fluid at 600° F (588 K) and a thrust load of 5800 pounds (25 800 N) under a low oxygen environment (less than 0.1 percent by volume). Within 8 hours running time, considerable ball wear and/or surface distress was observed. These tests indicated that at a maximum Hertz stress of 323 000 psi (223 000 N/cm²) mixed lubrication existed with boundary lubrication being the predominant mode. As a result, the thrust load was lowered to 4365 pounds (19 416 N), which produced a maximum Hertz stress on the inner races of 295 000 psi (203 000 N/cm²). Under this load condition, mixed lubrication existed with EHD lubrication being predominant. Some diametrical ball wear was apparent as evidenced by a decrease in ball diameter of 0.0005 to 0.002 inch (0.0013 to 0.0051 cm) within 500 hours of operation.

The fatigue results with this lubricant are presented in figure 8 and are summarized in table IV. The failure index for these tests is four failures out of 17 tests. For comparative purposes, the AFBMA-predicted (catalog) life is also presented.

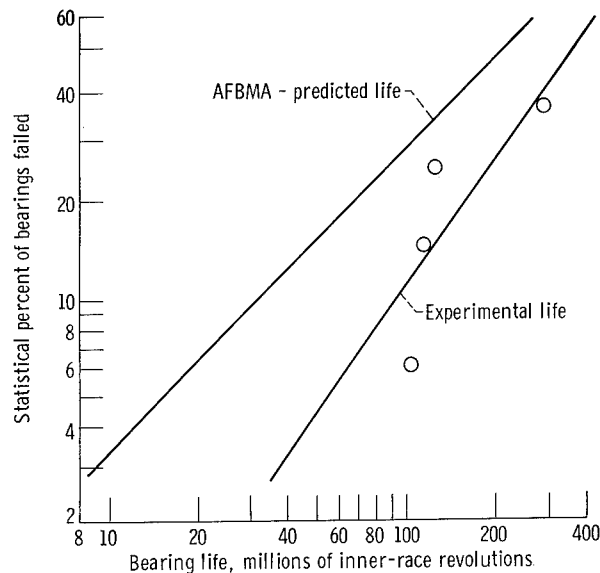


Figure 8. - Rolling-element fatigue life of 120-mm bore angular-contact ball bearings with fluorocarbon fluid. Material, AISI M-50 steel; thrust load, 4365 pounds (19 416 N); speed, 12 000 rpm; temperature, 600° F (588 K); low oxygen environment; failure index, 4 out of 17.

For most of the bearings, some glazing of the race surfaces was apparent. In some instances, this glazing was accompanied by superficial pitting. This condition is indicative of a very thin EHD film with frequent asperity contacts occurring (ref. 1).

In those tests that exceeded 400 hours (288×10^6 inner-race revolutions), there was some evidence to indicate that the fluid chemically attacked the AISI M-50 bearing material. However, the magnitude of this effect was minimal under the low oxygen environment and would not seriously influence bearing life or performance. No corrosion was noted on the nickel-base alloy cages.

The fluid exhibited a significant increase in viscosity after a relatively short period of time, and a more gradual but constant increase followed as time progressed (fig. 9).

The foaming problem encountered with the synthetic paraffinic oil was not present with this fluid. However, the high volatility of this batch of fluorocarbon oil necessitated the addition of about $1\frac{1}{2}$ gallons (5700 cm^3) of makeup fluid during each 100 hours of test. (This fluid was an early batch with a wide range of molecular weights. In a polymeric fluid where the light ends are volatilized, such as in this case, the heavier, high-viscosity ends remain to produce fluid with increased viscosity.)

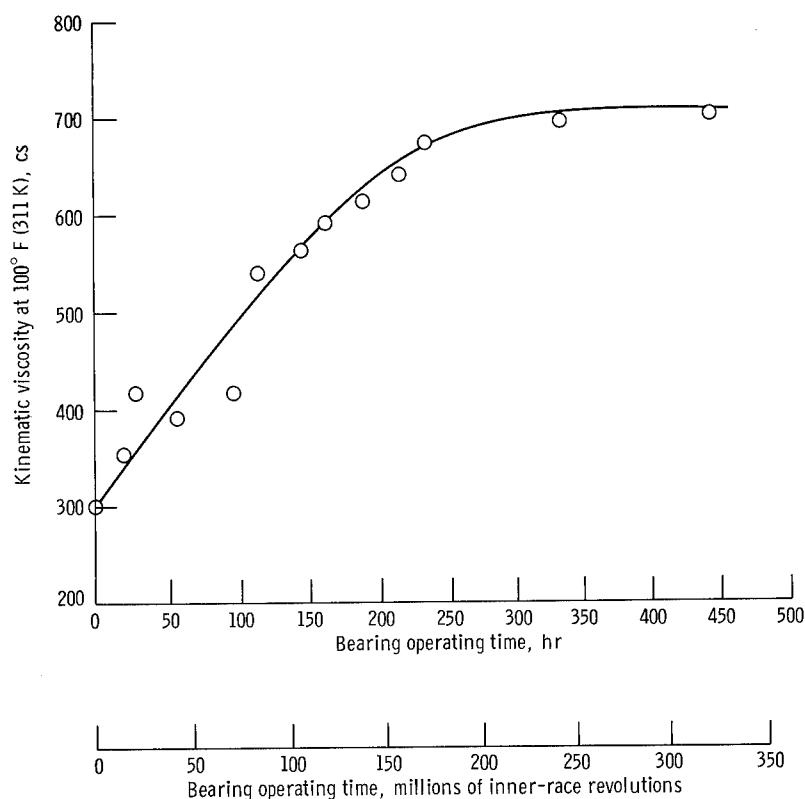


Figure 9. - Effect of thermal exposure and mechanical stressing on viscosity of fluorocarbon fluid. Bearing operating temperature, 600° F (588 K); low oxygen environment; lubricant flow through bearing, $1\frac{1}{2}$ gallons per minute ($95 \text{ cm}^3/\text{sec}$); 50 percent makeup fluid added per 72×10^6 inner-race revolutions.

Polyphenyl Ether

A preliminary two-bearing test was conducted with the 5P4E polyphenyl ether at 600° F (588 K) under a low oxygen environment (less than 0.1 percent by volume) at a bearing thrust load of 5800 pounds (25 800 N), which produced a maximum Hertz stress of 323 000 psi ($223\,000 \text{ N/cm}^2$) on the inner race. The test was shut down after 2 hours of operation because of the inability of the bearings to stabilize temperature. A very high noise level accompanied bearing operation. Disassembly of the bearings showed considerable ball wear, track damage, and a catastrophic cage failure on one of the bearings.

A second preliminary test, also conducted under a low oxygen environment, was initiated at a thrust load of 4365 pounds (19 416 N) which produced a maximum Hertz stress of 295 000 psi ($203\,000 \text{ N/cm}^2$). The test was shut down after 45 minutes of operation, again because of an inability to maintain the temperature of the bearings at 600° F (588 K). The outer-race temperature continued to climb above 600° F (588 K). Disassembly of the bearings showed fatigue failures on the inner race of one bearing and on the outer race of

the other. In addition, surface glazing of the races was apparent. Examples of these failures are shown in figure 10. Furthermore, the balls from both these bearings exhibited severe wear. The cages from both bearings were relatively undamaged. The appearance of the bearing surfaces indicated a very thin EHD film with frequent asperity contacts occurring.

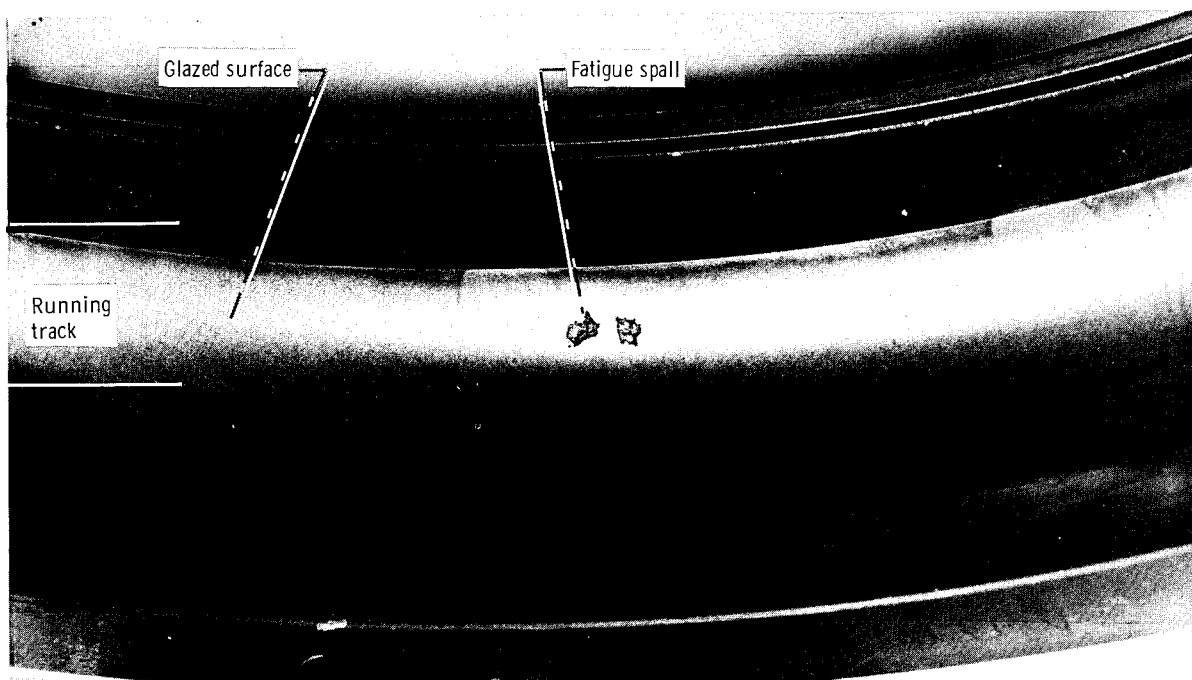
Based on previous research (refs. 1 and 2), a third preliminary test was conducted in an air atmosphere under conditions otherwise duplicating those of the second test. This third preliminary test was run for 120 hours, at which point the bearings were disassembled and inspected. The bearing components were in good condition with no apparent evidence of wear or fatigue damage. In addition, the bearing temperatures could be stabilized and controlled at approximately 600° F (588 K). However, some surface glazing was apparent. It was thus concluded that, under the marginal EHD conditions present with the 5P4E polyphenyl ether at 600° F (588 K), the protective oxides formed by the presence of oxygen in the air environment were sufficient to prevent gross surface damage. These results substantiate that bearings lubricated with polyphenyl ether cannot be successfully operated for long periods in a low oxygen environment under these test conditions.

On the basis of the results of the third preliminary test, a series of 26 bearings was tested in an air environment at the thrust load of 4365 pounds (19 416 N). Despite the presence of an air environment and the reduced load, most of the tests with the polyphenyl ether had to be suspended because of high ball wear. In one bearing, wear reduced the ball diameter as much as 0.025 inch (0.064 cm). In tests running from 2 to 65 hours, the average reduction in ball diameter was approximately 0.001 inch (0.0025 cm). However, 10 bearings ran for over 450 hours (325×10^6 inner-race revolutions) with minimal wear. The ball diameters on these bearings were reduced by not more than 0.0003 inch (0.0008 cm). A representative bearing race and balls are shown in figure 11. On all the bearings tested, glazing of the contacting surfaces was present. However, with the long-lived bearings, no micropitting of the surface accompanied the glazing.

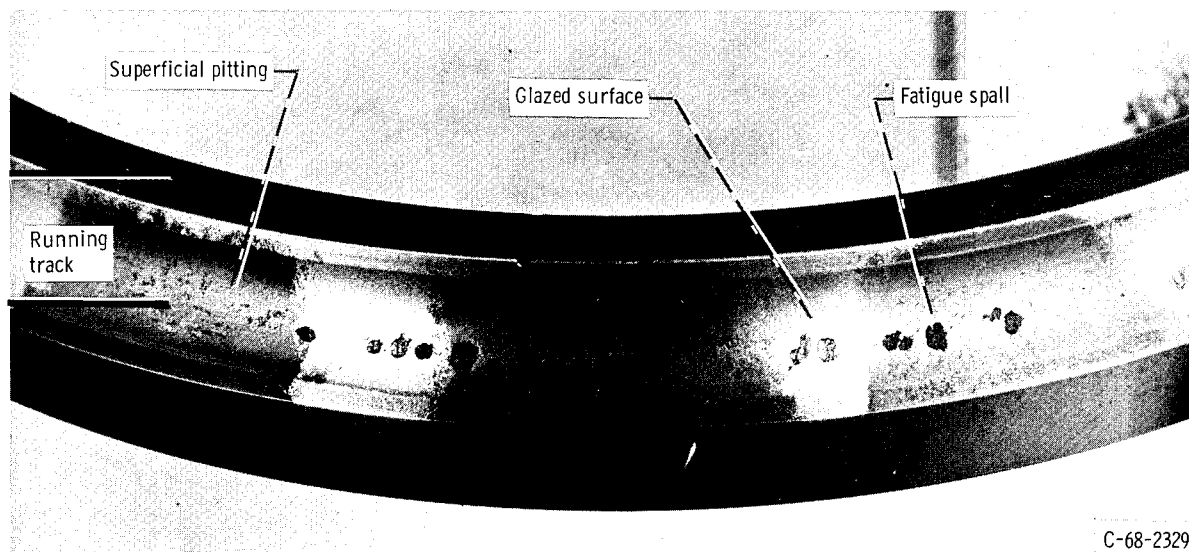
The results of these tests indicated that mixed lubrication occurred with the polyphenyl ether and that boundary lubrication was predominant. After a relatively short running time, a stable suspension of wear particles manifested themselves in the fluid. These particles can, and probably do, act as an abrasive which can aggravate the existing wear process.

A representative fatigue spall that occurred with the polyphenyl ether is shown in figure 12. For the failed bearings, the pitting was more shallow than that for the bearings run with the synthetic paraffinic oil. This is believed caused by high tangential forces due to a greater amount of asperity contact during operation.

Of the 26 bearings tested, only two failed by fatigue. This is an insufficient number of failures to permit drawing a Weibull line. However, a rough estimate of bearing life with this fluid was made in figure 13 on the basis of the fatigue data. This figure also



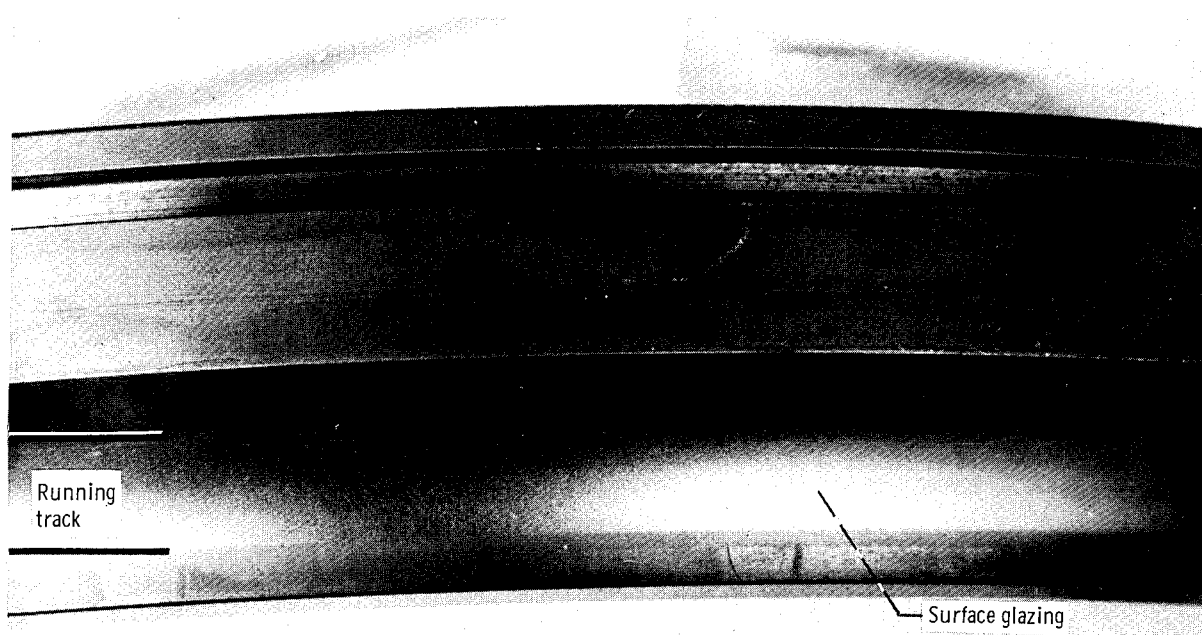
(a) Outer race.



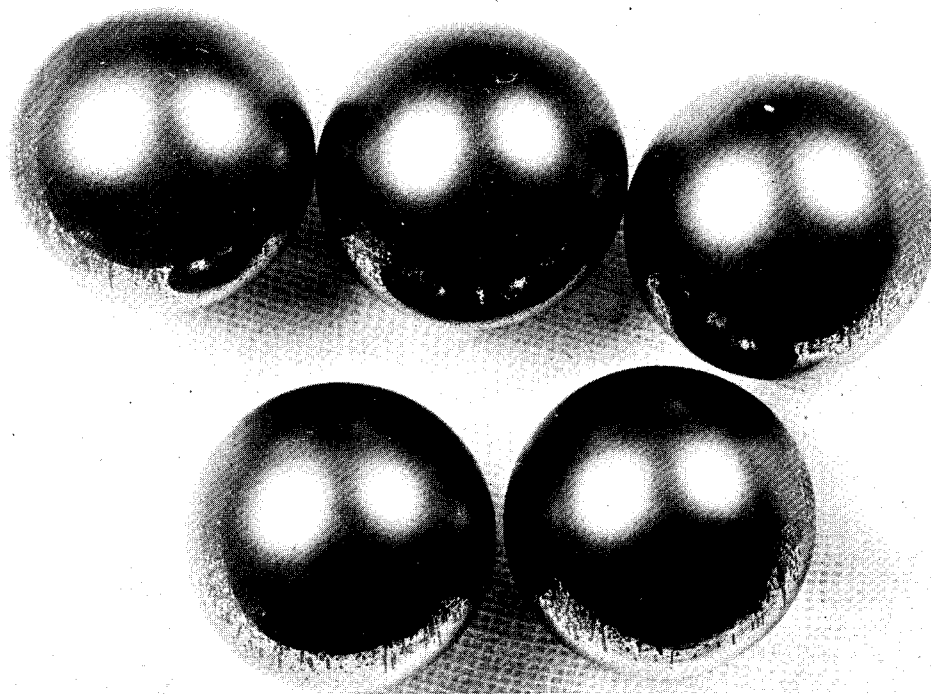
C-68-2329

(b) Inner race.

Figure 10. - Representative bearing failures for 120-mm angular-contact ball bearings run with polyphenyl ether. Material, AISI M-50 steel; thrust load, 4365 pounds (19 416 N); speed, 12 000 rpm; temperature, 600° F (588 K); low-oxygen environment; running time, 0.75 hour (0.54×10^6 inner-race revolutions).



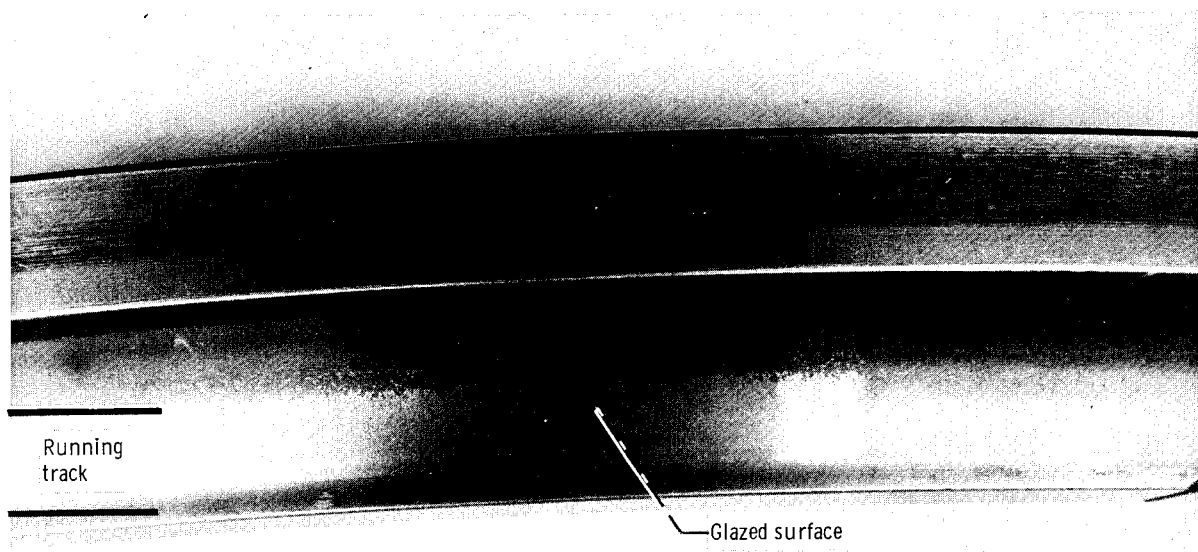
(a) Inner race.



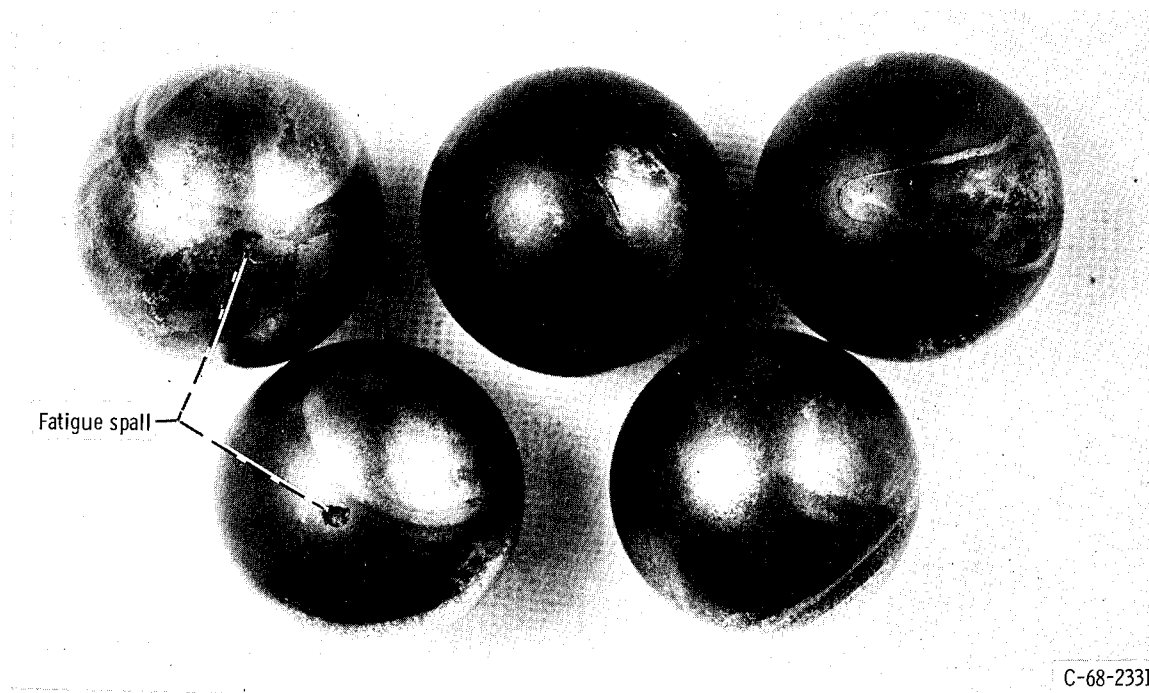
C-68-2330

(b) Balls.

Figure 11. - Unfailed 120-mm angular-contact ball bearing run with polyphenyl ether lubricant. Material, AISI M-50 steel; thrust load, 4365 pounds (19 416 N); speed, 12 000 rpm, temperature, 600° F (588 K); air environment; running time, 500 hours (350×10^6 inner-race revolutions).



(a) Inner race.



(b) Balls.

Figure 12. - Representative fatigue failure for 120-mm angular-contact ball bearings run with polyphenyl ether lubricant. Material, AISI M-50 steel; thrust load, 4365 pounds (19 416 N); speed, 12 000 rpm; temperature, 600° F; (588 K); air environment; running time, 13.2 hours (9.5×10^6 inner-race revolutions).

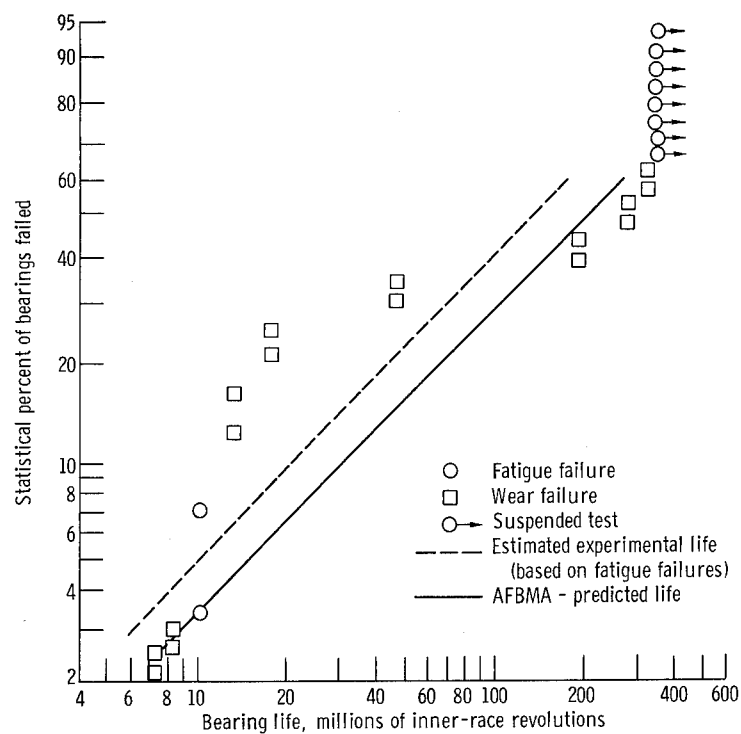


Figure 13. - Rolling-element fatigue life of 120-mm bore angular-contact ball bearings run with polyphenyl ether lubricant. Material AISI M-50 steel; thrust load, 4365 pounds (19 416 N); speed, 12 000 rpm; temperature, 600° F (588 K); air environment; failure index, 2 out of 26.

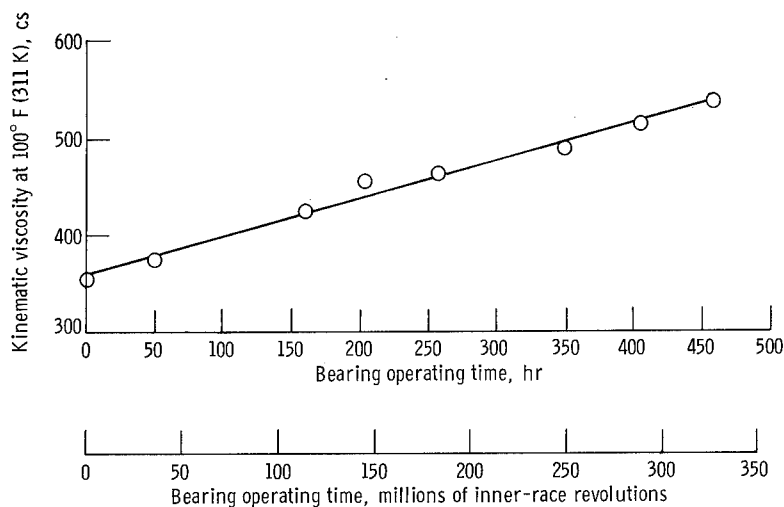


Figure 14. - Effect of thermal exposure and mechanical stressing on viscosity of polyphenyl ether lubricant. Bearing operating temperature, 600° F (588 K); low oxygen environment; lubricant flow rate through bearing, $1\frac{1}{2}$ gallons per minute (95 cm³/sec); 50 percent makeup fluid added per 58×10^6 inner-race revolutions.

shows the failure distribution for the other bearings tested. The AFBMA-predicted (catalog) life distribution is given for comparison purposes. These lives are summarized in table IV.

With the polyphenyl ether, oil consumption was quite high. It was estimated that at least 30 percent of the oil was lost by evaporation during a 24-hour operating period. Because of this evaporation, fluid had to be periodically added to the sump. Consequently, the periodic diluent effect of these fluid additions made it difficult to establish the true viscosity change with time. However, the 5P4E polyphenyl ether did show an overall increase in viscosity with time. A representative viscosity-time curve is shown in figure 14.

GENERAL COMMENTS

Based on the results of the tests with the three lubricants at 600° F (588 K), a bar chart of relative life at an equivalent thrust load is presented in figure 15. For the same

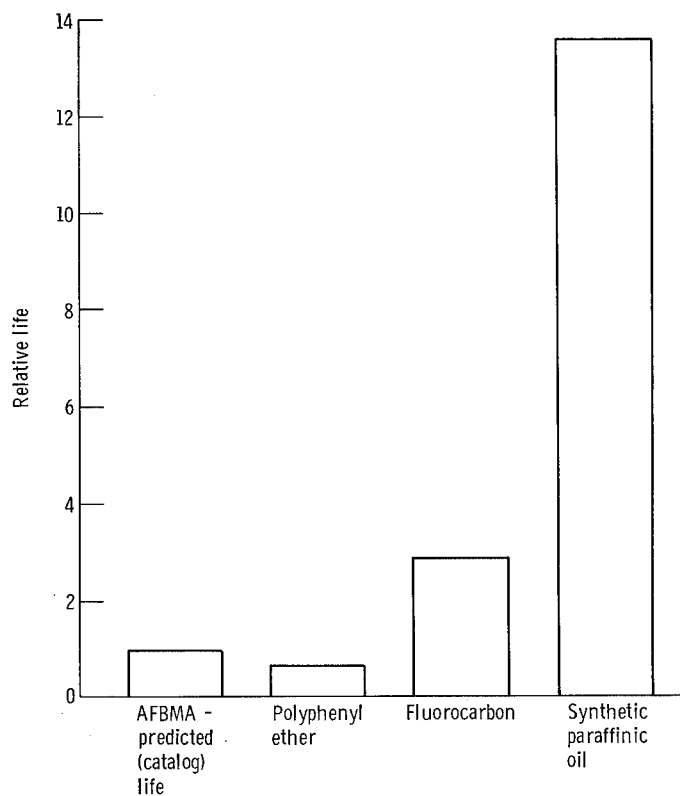


Figure 15. - Comparison of relative bearing lives with three high-temperature lubricants run at equivalent thrust load at 600° F (588 K) with AFBMA - predicted (catalog) life at room temperature.

bearings it can be seen that, for the synthetic paraffinic oil, the 10-percent life is approximately 14 times that of the AFBMA prediction. The fluorocarbon produced a life approximately three times that of AFBMA. On the other hand, the polyphenyl ether produced a bearing life less than the AFBMA-predicted life.

Some surface glazing was apparent with the synthetic paraffinic oil; however, no measurable wear occurred on the bearings. In most cases, the original grinding marks were visible after 500 hours of operation, which would indicate that EHD lubrication was predominant at a maximum Hertz stress of approximately 323 000 psi ($223\,000\text{ N/cm}^2$) at a 600° F (588 K) bearing temperature.

Small changes in lubricant viscosity, observed after exposure of the synthetic paraffinic oil for a period of 500 hours at high temperature, indicated a long-time thermal stability. A nitrogen (low oxygen) atmosphere was used with the synthetic paraffinic oil in the tests reported to prevent excessive oxidation from occurring at 600° F (588 K). At temperatures above 500° F (533 K), some type of partial inerting would be necessary even with an oxidation inhibitor in order for the fluid to be viable. The benefits provided by use of this lubricant would be excellent bearing performance and extremely long life at temperatures significantly above those seen today. Bearing life can be expected to at least equal that of bearings in current turbine engines lubricated with ester-base lubricants.

The polyphenyl ether appears to require an air environment in order to be an effective lubricant. The ability to function in air seems to be an important factor in using this fluid. This is established by the data presented herein and those of references 1 and 2. However, even in an air environment at a stress of 295 000 psi ($203\,000\text{ N/cm}^2$), the fluid appears to have questionable EHD capability and thus is marginal at the elevated temperature. At a lower stress level, it is conceivable that sufficient EHD lubrication can be attained to provide adequate bearing life if wear can be tolerated. This wear process manifests itself by a loss in bearing tolerances and a stable suspension of wear particles in the fluid.

On the basis of the chemical makeup of the three fluids, the fluorocarbon material has the greatest potential as an extremely high-temperature lubricant, that is, at temperatures greater than 600° F (588 K). Conceivably, the most important property of this fluid is its inherent fire resistance. The autoignition temperature of this lubricant has not been established, although it is known to be above 1300° F (978 K). The fire safety margin of the fluorocarbon is of considerable interest to engine designers; however, its corrosive aspects would tend to make it difficult to apply in existing lubricating systems. This problem should not be a major one in advanced engines where the lubricating system can be designed with materials which will accommodate the potential corrosive effects. In the reported tests, the fluorocarbon did not exhibit the severe corrosive effects under a low oxygen environment that had been anticipated. The overall damage to both

the test facility or the test bearings was within tolerable limits.

Although the fluorocarbon gave fatigue lives which exceeded AFBMA predictions, the performance of this fluid was not consistent but exhibited a mixed lubrication mode. The synthetic paraffinic oil and the polyphenyl ether also exhibited mixed lubrication modes. However, for the polyphenyl ether, boundary lubrication was the predominant mode; while for the synthetic paraffinic oil, EHD lubrication was predominant. For the fluorocarbon, both modes were apparent, although not predictable; that is, in one test there appeared to be predominantly EHD lubrication and in another, boundary lubrication.

The high bearing stabilization temperature with the fluorocarbon presents an extremely serious problem with respect to its application to an engine. If this fluid were to be used, it would be necessary to re-evaluate the entire cooling system of an aircraft turbine engine.

SUMMARY OF RESULTS

Groups of 120-mm bore angular-contact ball bearings made from AISI M-50 steel were fatigue tested with a synthetic paraffinic oil, a fluorocarbon, and a 5P4E polyphenyl ether. Test conditions included an outer-race temperature of 600° F (588 K), a speed of 12 000 rpm, and a thrust load of 4365 or 5800 pounds (19 416 or 25 800 N) producing a maximum Hertz stress of 295 000 or 323 000 psi (203 000 or 223 000 N/cm²) on the bearing inner race. The following results were obtained:

1. Under a low oxygen environment and at a maximum Hertz stress of 323 000 psi (223 000 N/cm²), the synthetic paraffinic oil had a life approximately 14 times the AFBMA-predicted (catalog) life. The fluid provided adequate elastohydrodynamic lubrication whereby no measurable wear occurred.

2. The fluorocarbon fluid under a low oxygen environment and at a maximum Hertz stress of 295 000 psi (203 000 N/cm²) provided a bearing life approximately three times the AFBMA-predicted (catalog) life. The fluid did not provide consistent lubrication results. The predominant lubrication mode ranged from boundary to elastohydrodynamic.

3. With the 5P4E polyphenyl ether in an air atmosphere and at a maximum Hertz stress of 295 000 psi (203 000 N/cm²), component wear and surface distress occurred, which indicated predominantly boundary lubrication. The life result was less than the AFBMA-predicted (catalog) life.

4. For the synthetic paraffinic oil and the fluorocarbon at 600⁰ F (588 K), bearing fatigue was subsurface initiated. For the polyphenyl ether, failure was caused mainly by surface distress, wear, and superficial pitting.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, July 1, 1968,
126-15-02-16-22.

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